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PRESSURE DROP AND HEAT TRANSFER IN A VERTICAL INTERNALLY HEATED ANNULUS

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Liu, Kuang-Ming

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by

Liu, Kuang-Ming

Lieutenant, Chinese Navy

Submitted in partial fulfillment of
the requirement for the degree of

MASTER OF SCIENCE

IN

MECHANICAL ENGINEERING

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Monterey, California

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ABSTRACT

The heat transfer and pressure drop of water flowing both upward and downward in an annulus were investigated. The data were obtained from a vertical narrow space annulus which was formed between 1.240 in. I.D. and 1.125 in. O.D. 99.9% copper tubes 142 in. long. By varying the annular fluid flow rate, the Reynolds number, based on an equivalent diameter, $(D_2 - D_1)$, covers a range from 150 to 10,000.

The pressure drop data were obtained for four different conditions: (1) isothermal runs with annular fluid flowing upward; (2) isothermal runs with annular fluid flowing downward; (3) internally heated runs with annular fluid flowing upward; and (4) internally heated runs with annular fluid flowing downward. The results from the four different conditions were practically the same when the average outer annulus wall temperature was employed to evaluate the fluid physical properties for internally heated runs. Within the laminar flow region, the values of friction factor are in excellent agreement with the theoretical predictions. Friction factors for Reynolds numbers greater than 2000 fall on the accepted empirical line for commercial pipes.

The heat transfer tests were conducted for two different flow conditions; one was for the annular fluid flowing upward, the other was for the annuler fluid flowing downward. These conditions were used in order to attempt to analyze the effects of natural convection. Steam heated water, circulating at about 10 ft/sec velocity through inner tube from top to bottom was used as heat source. Total heat transfer was based on the annulus side. Surface coefficients of heat transfer between the inner tube and annular fluid were determined by subtracting from the

overall resistance to heat transfer the sum of the resistances due to the inner hot water film and the metal wall. The present heat transfer data for the fluid flowing up is compared with the results of several other investigations. In general for the laminar region these results are lower and in the turbulent region they are somewhat higher.

The effect of natural convection on the heat transfer coefficients of this annulus is higher than that predicted for infinite width parallel plates. The effect of natural convection on the j-value is about 20% lower for annular fluid flowing down than flowing up within the laminar flow region.

The local heat transfer coefficient was determined in this investigation by assuming that the bulk temperature distribution was similar to the outer wall temperature distribution. Upon comparing the averaged value of the local coefficient computed in this manner with the average value as computed using the logarithmic mean temperature difference, discrepancies were noted.

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LIST OF SYMBOLS

English Letter Symbols

A	Area (ft^2)
C_p	Specific heat (Btu/lbm.F)
D	Diameter (ft)
De	$D_2 - D_1$, Equivalent Diameter (ft)
g	Acceleration of gravity (32.2 ft/sec^2)
G	$\dot{m}/A$ ($lbm/hr.\ ft^2$)
h	Convective coefficient of heat transfer ($Btu/hr.\ ft^2\,°F$)
h'	Natural convective coefficient of heat transfer ($Btu/hr.\ ft^2\,°F$)
k	Thermal Conductivity (Btu/hr. ft.°F)
L	Length (ft)
$\dot{m}$	Mass rate of flow (lbm/hr)
p	Pressure (lbf/ft^2)
Q	Heat transfer per unit time (Btu/hr.)
t	Temperature (°F)
u	Velocity (ft/sec)
x	Length (ft)

Greek Letter Symbols

Δ	Difference
μ	Viscosity (lbm/ft.sec)
ρ	Density (lbm/ft^3)
ϕ	Function
β	Coefficient of thermal expansion (1/°R)

Subscripts

b	Bulk value
c	Annulus fluid
d	Annular fluid flowing down
h	Pipe fluid
i	Inlet
LMD	Logarithmic mean difference
n	Average value
M	Metal
u	Annular fluid flowing up
o	Outlet
1	Inner annulus wall
2	Outer annulus wall
3	Inside wall of inner tube

Dimensionless Groups

Gr	Grashof number	$(g \beta \rho^2 \Delta t\, D_e^3/\mu^2)$
Nu	Nusselt number	$(h D_e/k)$
Pr	Prandtl number	$(C_p \mu/k)$
Re	Reynolds number	$(D_e G/\mu)$
St	Stanton number	$(h/G C_p)$
f	Friction Factor	$\frac{(\Delta P/\rho) D_e}{2(u^2/g) L}$

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1. Introduction:

Designers of heat transfer equipment have long needed heat transfer and pressure drop correlations which can be used for determining the coefficient for the case of an annulus space. Furthermore with the advent of nuclear reactors, special interest has also been focused on the heat transfer characteristics of water flowing in long internally heated annuli. Although a number of investigations, [1] , [2] , [3] , [4] *, have been made on the heat transfer in annulus space in recent years, the existing data are still sparse and no completely satisfactory correlation has been made, especially in the region of transition from laminar to turbulent flow.

It was the purpose of this investigation to obtain the correlations of pressure drop and heat transfer for water flowing in vertical annulus as coolant; and investigate the effects of natural convection for the change in direction of flow of the annular fluid. Steam heated hot water flowing down in the inner tube was used as the heat source. The annulus was made by two concentric copper tubes.

The Reynolds number range of this investigation was from 150 to 10,000 so as to cover a sufficient range of the laminar and transition regions.

* Number in brackets refers to the Bibliography

2. Description of Equipment.

Heat Exchanger. The heat exchanger proper consisted of a single vertical 1.125 inch O.D. x 0.995 inch I.D. 99.9% copper tube inside a 1.375 inch O.D. x 1.240 inch I.D. 99.9% copper tube 142 inches long. The annular diametric clearance was 0.115 inches and was maintained by 36 stainless steel rods, 0.0625 inches in diameter and 0.0575 inches high, fastened to the inner tube wall by solder. Detail of construction is shown in Figure 1 and essential dimensions are shown in Table 1.

Four mixing chambers for the inlet and outlet of fluids were made of 0.0625 inches copper sheet, 3 inches in diameter and 4 inches long with four baffles inside. Copper-constantan immersion thermocouples in the mixing chambers were used to measure the inlet and outlet bulk temperatures. A switch was built which enabled both absolute temperatures and temperature differences to be read. A schematic wiring diagram of the switching arrangement is shown in Figure 2. The circuit was designed to eliminate the interference from the circulating circuits existing between all the thermocouples. On the outer annulus tube 36 copper-constantan thermocouples were anchored by soldering in 1/32 inch holes drilled half way through the tube wall. These were placed in groups of three, equally spaced around a diameter, at 11 inch intervals down the length of the tube.

Pressure taps were drilled in the outer tube at a distance of 21 inches from each end giving a distance of 100 inches between taps. Both mercury and water manometers were used to measure pressure drop. For reading the small pressure differentials a water manometer was used and the mercury manometer was used for larger pressure differentials.

Auxiliary Equipment. The overall test assembly is shown in Figure 3. The hot water was heated within a large heating tank by about 20 psig steam which was reduced from the 120 psig saturated steam used in the school. The water was not changed during the investigation, therefore assuring a minimum of entrapped gasses and particles in the circuit. Fischer and Porter Flowrator meters were used to measure the flow rates for both hot and cold systems. These also served to indicate stability of the flow. The 36 outer wall thermocouple readings were taken with a Minneapolis-Honeywell multi-point recording potentiometer. These readings were recorded to a precision of $\pm$ 0.2°F. All other immersion thermocouples were read with a portable precision hand balance Rubicon potentiometer which was accurate to the $\pm$ 0.1°F.

The heat exchanger proper was wrapped with a 2 inch width asbestos cloth and then covered with a 1.5 inch thick pipe lagging.

3. Procedure:

Friction: To obtain isothermal friction data, pressure drop measurements were made of the fluid flowing in the annular space at various flow rates. The pressure taps were so located to give an entrance length of 180 equivalent diameters. Pressure differences of less than 66 in. of water were determined with a water manometer. For pressure differences of 66 in. of water or more a mercury manometer was used. When there was no hot water flowing through the inner tube the temperature of the fluid flowing in the annular space for most of the runs was essentially room temperature. When there was hot water flowing through the inner tube, the temperature of the fluid flowing in the annular space varied along the tube. Therefore, the temperatures of the outer annular wall were recorded for each run. The average outer wall temperature has been used to evaluate the fluid properties. The flow meters were calibrated before the starting of these tests. At low flow rates the flow meter measurements varied as much as $\pm$ 2% compared to the flow rate obtained by weighing, but for the high flow rates the variation was less than $\pm$ 1%.

Heat Transfer: In order to maintain nearly constant conditions in the inner tube, the inlet temperature of the hot water was held constant at about 102°F. This was readily accomplished by regulating the flow of steam to the heating tank. The water velocity in the inner tube was maintained constant at a value of around 10 ft./sec., corresponding to a Reynolds number of about 116,000. As a precaution against liberation of inert gases, the water in the hot water flow passage was circulated by a large capacity, 30 gal./min., pump.

For the annulus flow system, the flow rates were varied from 570 lbm/hr to 4000 lbm/hr, corresponding to Reynolds numbers of 200 to 10,000, thus providing data in both the laminar and transition regions.

In order to investigate the effect of natural convection on the coefficient of heat transfer, runs were first made with the cooling water flowing from bottom to top; then, after merely changing the cooling water flow direction, all the runs were repeated.

Data for the runs were taken after five to ten minutes of steady state operation of the system. Steady state conditions were denoted by the constancy of flow rates and temperatures and were usually reached within an hour after changes in operating conditions had been made.

4. Results:

Friction: Isothermal friction data are tabulated in Tables 2 and 3, the results shown in Figures 4 and 5. Tables 4 and 5 and Figures 6 and 7 show the data and results for the internally heated conditions. Tables 2 and 4 and Figures 4 and 6 are for the cooling water flow upward. Tables 3 and 5 and Figures 5 and 7 are for the cooling water flow downward. The average outer wall temperature was used in evaluating the fluid properties for internally heated runs. For the isothermal runs either bulk temperature or average outer wall temperature were used for evaluating the properties because they were essentially the same.

Heat Transfer: The heat transfer results are shown in Figure 8 and tabulated in Tables 6 and 7 for the cooling water flow upward and downward respectively. Tables 6 and 7 also present the temperatures, flow rate and all other data used to calculate the system parameters as well as the parameters themselves.

The overall heat transfer coefficient, U_j and convective coefficient of heat transfer between the inner tube and the annular fluid, h_1, are based on the outside surface area of the inner tube. The latter coefficient was computed by subtracting from the overall heat transfer resistance, 1/U, the sum of the water film resistance for the inner tube and the resistance due to the copper wall, each adjusted for its appropriate area as follows:

$$\frac{1}{h_1} = \frac{1}{U} - \frac{L_M D_1}{k_M D_{av.}} - \frac{D_1}{h_3 D_3} \qquad (1)$$

Evaluation of Inner Tube Coefficients of Heat Transfer: Since the velocity and the average temperature of the hot water in the inner

tube remained essentially constant for all runs, these constant values were employed in calculating the heat transfer coefficient of the water flowing through the inside tube. The average hot water temperature was taken to be 102.6°F and the mass flow rate as 12,600 lbm/hr. Using these values, the film coefficient for heat transfer inside tubes in turbulent flow was calculated using equation:

$$St = 0.023\ Pr^{0.8}\ Re^{0.4}$$

quoted by McAdams [9] as, h_3 = 2055 BTU/hr. sq. ft. F. Thus, Equation (1) after substitution of appropriate values, becomes:

$$\frac{1}{h_i} = \frac{1}{U} - 0.000616\,25 \qquad (2)$$

All values of h_1 in Tables 6 and 7 were read from the Figure 9 which was plotted by using the Equation (2).

Mean Temperature Difference: The logarithmic mean temperature difference was utilized to calculate the overall heat transfer coefficient, U.

From the definition of the logarithmic mean temperature:

$$\Delta t_{LMD} = \frac{\Delta t_a - \Delta t_b}{\ln \frac{\Delta t_a}{\Delta t_b}} \qquad (3)$$

Where

$$\Delta t_a = t_{ho} - t_{ci}$$

$$\Delta t_b = t_{hi} - t_{co}$$

for cooling water flowing up

and $\Delta t_a = t_{hi} - t_{ci}$

$\Delta t_b = t_{ho} - t_{co}$

for cooling water flowing down

The temperatures, t_{ho}, t_{hi}, t_{co}, t_{ci}, and the difference of them; such as, $t_{ho} - t_{ci}$, $t_{hi} - t_{co}$, for cooling water flowing upward, and the $t_{ho} - t_{co}$, $t_{hi} - t_{ci}$, for cooling water flowing downward, were observed individually. The results showed the directly observed differences were not the same as the subtracted values, especially for low flow rate runs. In order to get the better value of Δt_{LMD}, the arithmetic mean value of the directly observed difference and the subtracted differences were taken for Δt_a and Δt_b as tabulated in the Tables 6a and 7a.

The Inner Wall Temperatures of Annulus, t_{1n}: The average inner annulus wall temperatures were found from the energy rate equations:

$$t_{bh} - t_{3n} = \frac{Q_c}{A_3 h_3} \tag{4}$$

$$t_{3n} - t_{1n} = Q_c \frac{\ln \frac{r_1}{r_3}}{2\pi k L} \tag{5}$$

Adding Equations (4) and (5) together gives:

$$t_{1n} = t_{bh} - Q_c \left(\frac{1}{A_3 h_3} + \frac{\ln \frac{r_1}{r_3}}{2\pi k L} \right) \tag{6}$$

or $$t_{1n} = \frac{t_{hi} + t_{ho}}{2} - Q_c \left(\frac{1}{A_3 h_3} + \frac{\ln \frac{r_1}{r_3}}{2\pi k L} \right) \tag{7}$$

After substitution of appropriate constant values (take 110 BTU/hr.ft.F as the value of k for 99.9% copper tube wall), Equation (7) becomes:

$$t_{1n} = \frac{t_{hi} + t_{ho}}{2} - 0.00017664\, Q_c \tag{8}$$

All values of t_{1n} in Tables 6a and 7a were calculated by using Equation (8).

The Outer Wall Temperatures of Annulus, t_{2n}: Temperature readings of the 36 outer wall thermocouples which were anchored in groups of three and spaced 11 inches apart were recorded for each run. After taking the arthematic mean value of the three readings in every location, the temperature distribution along the outer wall was plotted as shown for a typical run in Figure 10. The average outer annulus wall temperatures for each run were found by measuring the area under the curve by using a planimeter.

The ratio of average outer wall temperature to bulk temperature of annular fluid versus Reynolds number was plotted and is shown in Figure 11. This average temperature was used to evaluate the fluid properties.

Local Heat Transfer Coefficients: A method for determining the local heat transfer coefficients was attempted for five selected runs of the annular fluid flowing up and flowing down, corresponding to about the same Reynolds numbers. This method is outlined as follows:

1. Plot the outer wall temperature and hot water bulk temperature versus the tube length. A linear distribution was assumed for the hot water bulk temperature distribution.

2. With the observed inlet and outlet bulk temperatures of the cooling water, plot an assumed cold water bulk temperature distribution curve similar in shape to the outer wall temperature curve.

3. After estimating the cold water bulk temperature distribution, the cold water bulk temperature at each location, t_{bcx}, (these locations were 11 inches apart) was read from the estimated curve. The local heat transferred, q_x, between any two adjacent locations was calculated by using the bulk temperature difference between the appropriate locations

and the mass flow rate of the fluid.

4. The local inner annulus wall temperature between two adjacent locations was calculated similar to the method used to find the inner annulus wall temperature as given by Equation (6), which when the appropriate values are substituted yields

$$t_{bhx} - t_{1x} = 0.002191\, q_x \tag{9}$$

By Equation (9) the local inner annulus wall temperature variation was plotted versus tube length, and the local temperature difference between inner wall, t_{1x}, and cold water bulk, t_{bcx}, was determined from the plot.

5. The local heat transfer coefficients at each location were then computed by the relation of

$$h_x = \frac{q_x}{(t_1 - t_{bc})_x A_{1x}} \tag{10}$$

where A_{1x} is the area of inner annulus wall between two adjacent locations and is equal to 0.270 ft^2. Thus,

$$h_x = \frac{q_x}{(t_1 - t_{bc})_x\, 0.270}$$

The detailed calculations for one run are presented in the Appendix.

6. An average of the local heat transfer coefficients was obtained by planimetering the area. These average values are compared with those foun by the logarithmic mean temperature difference and are shown in Figure 13.

5. Discussion.

Friction: The isothermal and internally heated friction data for both the conditions of the cooling water flowing upward and downward are plotted as friction factor versus Reynolds number based on the equivalent diameter, D_e, in Figures 4, 5, 6, and 7. The change from laminar to turbulent flow for all these flow conditions is seen to occur approximately at a Reynolds number of 2000 which corresponds to the critical value for transition of flow inside round tubes. As it is desirable for comparison purposes to have the critical value of Reynolds number which conforms with that of flow in round tube, it was decided the equivalent diameter was the preferred characteristic defining length. It was also found that to correlate the data for the internally heated runs in the laminar Reynolds number range, it was necessary to follow the method proposed by Colburn [10] and use a "special" temperature for the evaluation of the fluid properties. While for Colburn this special temperature was found to be the arithmetic average of the bulk temperature, for the present case the special temperature was found to be the average temperature of the outer annulus wall. Figure 11 shows the ratio of the average annulus outer wall temperature to the arithmetic average bulk temperature versus Reynolds number. From Figure 11 it is seen that above a Reynolds number of 2000, the present method of determining the "special" temperature and that of Colburn's are essentially the same.

As shown by Knudsen and Katz [7] besides the usual relations for the

friction factor,

$$f = \frac{(\Delta P/\rho)\, D_e}{2\,(u^2/g)\, L} \tag{11}$$

and Reynolds number,

$$Re = \frac{G\, D_e}{\mu} \tag{12}$$

for flow of fluids in an annulus an additional geometry parameter,

$$\phi\left(\frac{D_1}{D_2}\right) = \frac{\left(1 - \frac{D_1}{D_2}\right)^2}{1 + \left(\frac{D_1}{D_2}\right)^2 + \frac{\left[1 - \left(\frac{D_1}{D_2}\right)^2\right]}{\ln\left(\frac{D_1}{D_2}\right)}} \tag{13}$$

is needed. The value of function in Equation (13) is equal to unity for D_1/D_2 equal to zero, and approaches a value of 1.5 as D_1/D_2 approaches 1.0. For this particular annulus, where D_1/D_2 is equal to 0.906, the value of the function will be 1.499. The functional relationship between the system parameters for laminar flow is theoretically found to be

$$f = \frac{16}{Re}\,\phi\left(\frac{D_1}{D_2}\right)$$

or

$$\frac{f}{\phi} = \frac{16}{Re} \tag{14}$$

The agreement of these data with theory is seen in Figures 4, 5, 6, and 7, in the region of viscous flow.

For Reynolds numbers greater than 2000, all the data of these four different conditions were plotted and correlated as one equation:

$$\frac{f}{\phi} = 0.0366\,(Re)^{-0.20} \tag{15}$$

Equation (15) agrees with Colburn's[2] correlation very well, except the fluid properites are evaluated with the average outer annulus wall

temperatures for internally heated runs.

Average Heat Transfer Coefficients: The correlation was made for all runs of cooling water flowing up and down, based on the Colburn [10] analogy of

$$j = St(P_r)^{\frac{2}{3}} \left(\frac{\mu_{in}}{\mu_{zn}}\right)^{0.14} = \phi(Re)$$

with the fluid properties evaluated at the average outer annulus wall temperature. Although the present data does not cover a sufficient variation in temperature gradient to show any effect of the term, $(\mu_{in}/\mu_{zn})^{0.14}$, it is still used as an approximation.

The least mean squares method was employed to find the best slope of the curve in the laminar region. It was found to be -0.4430 for cold water flowing up, and -0.4694 for the cold water flowing down. In order to simplify the application of these results, the slope of -0.5 was used for both flow conditions. Thus in the laminar flow region, for cooling water flowing upward, the correlation is,

$$j_u = St\,(P_r)^{\frac{2}{3}} \left(\frac{\mu_{in}}{\mu_{zn}}\right)^{0.14} = 0.1320\,(Re)^{-\frac{1}{2}} \qquad (16)$$

and for cooling water flowing downward, the correlation is,

$$j_d = St\,(P_r)^{\frac{2}{3}} \left(\frac{\mu_{in}}{\mu_{zn}}\right)^{0.14} = 0.1065\,(Re)^{-\frac{1}{2}} \qquad (17)$$

Comparing the results of Equation (16) with those of Burt, [8] who used a similar annulus with upward flow where he found,

$$j_u = 0.105\,(Re)^{-\frac{1}{2}} \qquad (18)$$

It is seen that the exponent of the Reynolds number and, therefore, the slope of the curve is the same, but the average value of these results is about 20% higher. The lower value correlated by Burt is believed due to the faulty installation of his inner tube mounting.

According to Jacob and Rees, [4] :

$$j_d = 2.68\,(Re)^{-1.0} \tag{19}$$

and to Norris and Streid, [11] :

$$j_d = 0.138\,(Re)^{-\frac{2}{3}} \tag{20}$$

As shown in Figure 12, it is clear that these relations are valid only at very low Reynolds numbers.

In Figure 8, the results for the two different cooling fluid flow directions are shown plotted together for comparison. When the Reynolds number is greater than 2000, there is an appearance of the transition "dip region". The curves approach coincidence at a Reynolds number of 10,000. The j-value at the point of Reynolds number 10,000 is almost the same as the value determined by Sieder and Tate, [6] in the turbulent region at that point. A simple equation could not be developed for the relationship within the transition region, $2{,}000 < Re < 10{,}000$.

Local Heat Transfer Coefficients: Ten selected runs, five for each cold fluid flow condition (up and down), were used to determine the local heat transfer coefficient distribution utilizing the calculation technique previously outlined. One set of detailed computations is shown in the Appendix.

The average local heat transfer coefficient, h_n, along the whole tube length, based on the measurement of the area under the local heat transfer coefficient distribution curve for the ten selected runs was also found. The ratios of the average local heat transfer coefficient, h_n, to the heat transfer coefficient based on logarithmic mean temperature difference, h_{LMD}, tabulated in Tables 7b and 8b, versus Reynolds number are shown in Figure 13.

Even though it is believed that the approximated cold fluid bulk temperature variation is the best estimation, its accuracy is doubtful. The value of local heat transfer coefficient, h_x, depends upon the local cold fluid bulk temperature, t_{bcx}, as shown in Equation (10).

$$h_x = \frac{q_x}{(t_1 - t_{bc})_x A_{ix}} \tag{10}$$

and the computation of the local heat transfer, q_x, is also based on the cold fluid bulk temperature difference between adjacent locations. At each location these temperatures were read directly from the estimated bulk temperature distribution curve; therefore, the scattering of h_x, shown in Figure 15, and the h_n/h_{LMD}, shown in Figure 13, could not be avoided.

Effect of Natural Convection: The difference between the results of the two flow conditions, cooling water flow up and down, was believed to be the natural convection effect on the fluid in annulus.

According to the general relation of natural convection,

$$Nu = \phi (Gr\, Pr) \frac{D_e}{L} \tag{21}$$

where $$Nu = \frac{h' D_e}{k}$$

and $$Gr = \frac{g \beta \rho^2 D_e^3}{\mu^2} (t_{in} - t_{zn})$$

The present data were correlated by assuming,

$$h' \equiv \frac{1}{2} (h_u - h_d) \tag{22}$$

From Equations (16) and (17);

$$h_u = 0.132\, Re^{-0.5}\, Pr^{-\frac{2}{3}}\, G\, C_p \left(\frac{\mu_{zn}}{\mu_{in}}\right)^{0.14} \tag{23}$$

and

$$h_d = 0.1065\, Re^{-0.5}\, Pr^{-\frac{2}{3}}\, G\, C_p \left(\frac{\mu_{zn}}{\mu_{in}}\right)^{0.14} \tag{24}$$

Substituting Equations (23) and (24) into Equation (22) and multiplying by De/k on both sides yields,

$$Nu = 0.01275\, Re^{0.5}\, Pr^{\frac{1}{3}} \left(\frac{\mu_{zn}}{\mu_{in}}\right)^{0.14} \tag{25}$$

This correlation was found to be,

$$Nu = 0.195\, (Gr\, Pr)\, \frac{D_e}{L} \tag{26}$$

as shown in Figure 14. The fluid properties were again evaluated with the average temperature of the outer annulus wall.

Comparing the results of Equation (26) with that for the infinite width parallel plates which were studied by Elenbaas [5], the slopes are shown to be exactly the same; but the value of the correlation constant is much higher than that for the infinite width parallel plates. This difference is believed to be caused by the assumption of Equation

(22). Hence further investigation is needed to establish the effect of natural convection on the heat transfer coefficient of the fluid flowing in annulus. For this experiment, in general, the effect of natural convection is that the value of the j-factor for fluid flowing down is about 20% lower than the value of the j-factor for fluid flowing up within the range of Reynolds number of 300 to 2000. The effect of natural convection is decreased and approaches zero for Reynolds number above 10,000.

Uncertainties: Using the uncertainty analysis proposed by Thrasher and Binder [12], the uncertainty interval for the present data were calculated to the nearest one-half percent as:

Quantity	Uncertainty Interval at 20 to 1 odds
Total heat transfer, Q_c	± 2.0%
Overall heat transfer coefficient, U	± 3.5%
Heat Transfer Coefficient, h	± 5.5%
j-factor	± 6.0%
Reynolds number, Re	± 3.0%
Friction Factor, f	± 4.0%

6. Conclusions:

1. For the case of the internally heated annulus, friction data results show that the proper temperature for evaluating the fluid physical properties is the average outer annulus wall temperature, while in the Reynolds number the proper length dimension to employ is the equivalent diameter, D_e.

2. The friction data in the laminar flow region were correlated as:

$$\frac{f}{\phi} = \frac{16}{Re} \tag{14}$$

which closely checks the theoretical relation. It is recommended that Equation (14) be employed for annuli in this region of flow.

3. The friction data in the transition and turbulent regions where Reynolds number was greater than 2000 were correlated as:

$$\frac{f}{\phi} = 0.0366\ Re^{-0.20} \tag{15}$$

which agrees with Colburn's results; therefore, this line can be used for other sizes of annuli.

4. The heat transfer results in the transition region, Reynolds number between 2,000 and 10,000 are higher by 2% at Reynolds number of 2,100 to 11% at Reynolds number of 10,000 than the Sieder and Tate's results, for flow inside tube with $\left(\frac{L}{D_1 + D_2}\right) = 60$.

5. The heat transfer data in the laminar region, Reynolds number less than 2,000 were correlated as:

$$J_H = 0.132\ Re^{-0.50} \tag{16}$$

and $j_d = 0.1665\ Re^{-0.5}$ (17)

For the fluid flowing upward, results almost coincide at Reynolds number of 2,000 and are about 32% lower at Reynolds number of 300 than the Sieder and Tate line which slope is -2/3.

Tests with a similar annulus were run by Burt in 1960. Because of faulty installation of the inner tube, his results are about 20% lower than these data, but with the same slope as these results. Hence, in general, for extremely narrow annuli the present curves, within the range of Reynolds number 300-10,000, are recommended.

6. The true local heat transfer coefficient variation can not be found until the bulk temperature variation of the cold fluid within the annulus is completely obtained.

7. The effect of natural convection on the heat transfer coefficients of this annulus is higher than that predicted for infinite width parallel plates. The effect of natural convection on the j-value is about 20% lower for annular fluid flowing down than flowing up within the laminar flow region.

APPENDIX

Computation of local heat transfer coefficient variation.

Following the method as outlined before, the detailed computations of local heat transfer coefficients for run number 14 of the cold water flowing upward is shown here.

1. From the observed data, t_{hi}=103.67°F, t_{ho}= 102.85°F, and the outer annulus wall temperatures, t_{2x}, (which are tabulated in the first column of Table 8), the hot water bulk and outer annulus wall temperature distribution curves were plotted as shown in the Figure 15. The linear distribution of the hot water bulk temperatures was assumed.

2. We know the cold bulk temperatures, t_{ci}= 76.88°F, and t_{co} = 98.36°F, of the annular fluid at the two ends. Using these two temperatures, we estimated the bulk temperature distribution curve of the annulur fluid. Then, the bulk temperature of the annular fluid at each location was read directly from the curve, these values are tabulated in the second column of Table 8.

3. Subtracting the annular fluid bulk temperatures between adjacent locations the difference of t_{bcx} between adjacent locations were found as shown in third column.

4. The local heat transfer between adjacent locations was found by the relation of

$$q_x = \dot{m} \Delta (t_{bc})_x \tag{22}$$

where the mass flow rate, $\dot{m}$, for this run is 0.1858 lbm/sec or 669 lbm/hr.

Then, Equation (22) becomes:

$$q_x = 669 \, (t_{bc})_x \qquad (23)$$

The values of qx calculated by Equation (23) are tabulated in the fourth column of Table 8 and plotted in Figure 15.

5. The local inner annular wall temperatures, t_{1x}, between locations were determined by Equation (9).

$$t_{bhx} - t_{1x} = 0.002192 \, q_x \qquad (9)$$

The values of $(t_{bhx} - t_{1x})$ are tabulated in column 5 and also plotted in Figure 15.

6. After establishing the curve of t_{1x}, the difference between t_{1x} and t_{bcx} for adjacent locations was measured directly from the two curves. The local heat transfer coefficient between each two locations was computed by the Equation (10)

$$h_x = \frac{q_x}{(t_1 - t_{bc})_x \, 0.270} \qquad (10)$$

The values of $(t_1 - t_{bc})x$ and h_x are tabulated in column 6 of Table 8. The local heat transfer coefficient distribtuion curve is shown in Figure 15.

7. The average value of the local heat transfer coefficient h_n, was found by measuring the area under the curve. Its value for this run is about 420 Btu/hr $ft^{2°}$ F.

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TABLE I.

APPARATUS DIMENSIONS

Inner Tube:

Effective tube length - 142 inches - 11.83 feet

Diameter (inside) - 0.995 inch - 0.08316 feet

Diameter (outside) - 1.125 in. -0.09375 ft.

Wall thickness - 0.0635 inch - 0.005291 ft.

Cross-Sectional Area (inside) - 0.005432 sq. ft.

Cross-Sectional Area (outside) - 0.006904 sq. ft.

Circumference (inside) - 0.2613 ft.

Circumference (outside) - 0.2946 ft.

Effective Heat Transfer Surface (inside) -3.0917 sq. ft.

Effective Heat Transfer Surface (outside) - 3.4866 sq. ft.

Jacket Tube:

Diameter (inside) - 1.240 in. - 0.1033 ft.

Diameter (outside) - 1.375 in. - 0.1146 ft.

Cross Sectional Area (inside) - 0.008389 sq. ft.

Circumference (inside) - 0.3247 ft.

Annulus:

Equivalent Diameter - 0.009583 ft.

Cross Sectional Area - 0.001485 sq. ft.

Effective Friction Length - 100 in. - 8.333 ft.

Table 2. Isothermal Friction Data, No Heat Source
(Cooling Water Flowing Upward)

w Rate ./sec.	Temp. F.	∆ H	u ft./sec.	f	f/ϕ	Re
01663	65.8	3.06"H_2O	0.1813	0.14350	0.09580	156
02500	65.7	4.50 "	0.2720	0.09400	0.06260	233
03350	65.5	6.19 "	0.3620	0.07300	0.04860	308
05000	63.4	10.25 "	0.5430	0.05370	0.03580	484
05910	68.1	11.33 "	0.6420	0.04440	0.02960	569
06660	68.0	12.38 "	0.7250	0.03640	0.02420	641
07510	68.0	13.94 "	0.8160	0.03235	0.02155	722
08200	68.0	15.19 "	0.8910	0.02955	0.01970	790
09370	68.0	17.50 "	1.0180	0.02610	0.01740	902
01110	67.8	20.94 "	1.2080	0.02210	0.01475	1060
03200	67.8	24.88 "	1.4350	0.01865	0.01241	1259
04850	67.8	28.19 "	1.6150	0.01663	0.01109	1418
06800	67.8	32.50 "	1.8280	0.01503	0.01000	1600
03750	67.8	36.80 "	2.0400	0.01365	0.00912	1789
00400	67.8	41.00 "	2.2200	0.01284	0.00856	1945
02900	67.8	47.00 "	2.4900	0.01169	0.00780	2185
04050	67.1	51.00 "	2.6150	0.01151	0.00768	2290
05950	67.1	58.50 "	2.8200	0.01135	0.00757	2470
07800	67.1	65.63	3.0200	0.01110	0.00741	2645
02650	67.0	6.92"Hg	3.5450	0.01068	0.00713	3105
03200	67.0	9.40 "	4.1500	0.01061	0.00708	3620
02700	64.5	11.40 "	4.5300	0.01080	0.00720	3810
01300	64.2	17.00 "	5.5300	0.01060	0.00707	4670
01600	65.2	22.80 "	6.7000	0.00987	0.00657	5710
09800	64.6	28.50 "	7.6000	0.00958	0.00638	6390
03200	64.0	35.80 "	8.5000	0.00964	0.00642	7110
09200	63.2	46.00 "	9.7000	0.00952	0.00634	8030

Table 4. Friction Data, Internally Heated
(Cooling Water Flowing Upward)

Flow Rate lbm./sec.	Temp.* F.	∆ H	u ft./sec.	f	f/ϕ	Re
0.01578	99.0	1.56"H_2O	0.1723	0.08250	0.05500	222
0.02445	98.9	2.88 "	0.2665	0.06260	0.04180	343
0.03275	97.3	4.19 "	0.3565	0.05110	0.03400	453
0.04110	96.1	5.31 "	0.4470	0.04120	0.02750	565
0.04940	94.8	6.19 "	0.5380	0.03310	0.02200	662
0.05810	93.5	7.88 "	0.6340	0.03040	0.02020	770
0.06630	93.1	9.13 "	0.7220	0.02700	0.01800	870
0.07470	92.3	10.06 "	0.8150	0.02350	0.01565	970
0.08130	91.3	10.81 "	0.8850	0.02140	0.01425	1048
0.09230	92.3	13.19 "	1.0040	0.02020	0.01350	1198
0.11050	90.0	15.83 "	1.2040	0.01700	0.01132	1429
0.12900	88.9	18.81 "	1.4020	0.01490	0.00993	1620
0.14650	88.0	21.50 "	1.5950	0.01310	0.00873	1820
0.16650	88.2	26.44 "	1.8100	0.01253	0.00837	2080
0.18410	88.1	29.88 "	2.0000	0.01160	0.00772	2295
0.20250	87.6	34.38 "	2.2000	0.01100	0.00734	2500
0.22100	87.2	40.13 "	2.4000	0.01080	0.00720	2725
0.23900	86.8	45.13 "	2.6000	0.01034	0.00689	2930
0.25900	86.5	52.56 "	2.8200	0.01022	0.00683	3165
0.27650	86.2	4.69"Hg	3.0100	0.01010	0.00675	3375
0.32650	85.3	6.45 "	3.5500	0.00998	0.00665	3940
0.37300	84.4	8.28 "	4.0500	0.00988	0.00657	4440
0.46600	83.6	12.68 "	5.0700	0.00964	0.00642	5500
0.55700	81.7	17.86 "	6.0600	0.00948	0.00632	6410
0.64000	80.5	23.71 "	6.9600	0.00954	0.00635	7250
0.73700	79.0	31.00 "	8.0100	0.00940	0.00627	8200
0.83200	77.8	39.10 "	9.0500	0.00930	0.00620	9100
0.91200	76.9	46.60 "	9.9100	0.00906	0.00605	9850

Temp*-The average temperatures of the outer wall.

Table 3. Isothermal Friction Data, No Heat Source
(Cooling Water Flowing Downward)

ow Rate m./sec.	Temp. F.	∆ H	u ft./ sec.	f	f/ϕ	Re
01668	66.9	3.00"H_2O	0.1812	0.14150	0.09440	158
02460	66.9	4.75 "	0.2675	0.10300	0.06870	234
03250	66.7	6.13 "	0.3535	0.07600	0.05070	308
04170	66.7	7.75 "	0.4530	0.05980	0.03990	396
05000	66.8	9.19 "	0.5440	0.04810	0.03210	475
05830	66.6	10.75 "	0.6340	0.04160	0.02770	554
06710	66.5	12.38 "	0.7300	0.03600	0.02400	635
07500	66.5	13.75 "	0.8150	0.03210	0.02140	710
08060	66.5	15.06 "	0.8760	0.03040	0.02020	763
09260	66.4	17.75 "	1.0100	0.02695	0.01800	875
11120	66.3	21.31 "	1.2100	0.02260	0.01510	1050
12990	66.3	24.50 "	1.4120	0.01885	0.01255	1225
14950	66.2	28.25 "	1.6250	0.01660	0.01108	1410
16680	66.1	31.63 "	1.8120	0.01495	0.00997	1570
18550	66.1	35.88 "	2.0200	0.01365	0.00910	1750
20400	66.1	39.75 "	2.2200	0.01250	0.00835	1920
22250	66.1	44.75 "	2.4200	0.01185	0.00790	2095
24100	66.1	49.75 "	2.6200	0.01125	0.00750	2270
26000	66.1	57.00 "	2.8250	0.01105	0.00737	2450
27850	66.1	63.25 "	3.0250	0.01071	0.00715	2620
29000	66.1	6.85"Hg	3.5850	0.01038	0.00692	3110
37150	66.1	8.50 "	4.0300	0.01020	0.00680	3500
41800	66.1	10.85 "	4.5500	0.01022	0.00682	3940
51500	66.2	16.10 "	5.6000	0.01004	0.00670	4850
60900	66.4	22.10 "	6.6100	0.00985	0.00657	5750
69200	66.6	28.55 "	7.5200	0.00980	0.00654	6570
78900	66.1	36.78 "	8.5700	0.00976	0.00651	7420
88800	66.8	46.10 "	9.6500	0.00970	0.00647	8430

Table 5. Friction Data, Internally Heated
(Cooling Water Flowing Downward)

Flow Rate lbm./sec.	Temp* F.	∆H	u ft./sec.	f	f/ϕ	Re
0.01663	98.4	2.25"H_2O	0.1818	0.10550	0.07050	232
0.02410	98.5	3.38 "	0.2630	0.07550	0.05040	336
0.03280	97.3	4.88 "	0.3580	0.05910	0.03940	452
0.04200	96.0	5.88 "	0.4580	0.04330	0.02890	570
0.05070	94.9	7.25 "	0.5530	0.03675	0.02450	680
0.05810	95.4	8.25 "	0.6350	0.03170	0.02110	785
0.06650	93.2	9.08 "	0.7260	0.02660	0.01775	878
0.07560	92.2	9.50 "	0.8230	0.02170	0.01450	985
0.08400	91.6	12.10 "	0.9160	0.02230	0.01490	1090
0.09220	90.1	13.08 "	1.0040	0.01980	0.01320	1170
0.11450	89.4	16.25 "	1.2500	0.01510	0.01008	1442
0.12920	88.1	18.50 "	1.4100	0.01445	0.00964	1610
0.14800	87.8	21.58 "	1.6150	0.01281	0.00855	1830
0.16650	87.5	24.88 "	1.8200	0.01162	0.00775	2060
0.18580	88.6	31.25 "	2.0270	0.01180	0.00785	2320
0.20350	87.5	34.44 "	2.2210	0.01083	0.00723	2515
0.22300	87.8	40.25 "	2.4400	0.01042	0.00695	2760
0.24050	88.5	46.38 "	2.6200	0.01088	0.00724	3000
0.26800	89.1	55.50 "	2.9200	0.01010	0.00674	3370
0.27700	87.8	4.75"Hg	3.0200	0.01018	0.00678	3440
0.32400	87.0	6.50 "	3.5400	0.01014	0.00677	3980
0.37000	86.8	8.30 "	4.0400	0.00992	0.00661	4520
0.46500	85.3	12.80 "	5.0750	0.00969	0.00645	5580
0.55500	84.3	18.00 "	6.0600	0.00956	0.00637	6600
0.66500	83.1	25.00 "	7.2600	0.00925	0.00616	7820
0.74800	81.7	32.50 "	8.1600	0.00954	0.00636	8600
0.82300	80.1	39.70 "	8.9800	0.00961	0.00641	9280

Temp*-The average temperatures of the outer wall.

Table [illegible]. Heat Transfer Data-Temperatures
(Cooling Water Flowing Upward)

No. of Run	t_{hi} F	t_{ho} F	t_{ci} F	t_{co} F	$\overline{t_{hi}-t_{ci}}^{*}$ F	$\overline{t_{ho}-t_{co}}^{*}$ F	Δt_a F	Δt_b F	Δt_{LMD} F	t_{1n} F	t_{2n} F
1	103.75	103.62	73.02	103.75	30.87	0.271	30.80	0.200	6.30	103.27	98.9
2	103.75	103.57	73.34	103.71	30.15	0.226	30.26	0.203	6.25	103.10	97.3
3	103.75	103.39	74.06	103.62	28.92	0.362	29.24	0.361	6.56	102.93	96.1
4	103.75	103.07	74.47	103.53	28.83	0.634	28.94	0.657	7.45	102.74	94.8
5	103.75	102.58	76.01	103.43	27.52	0.997	27.49	1.083	8.20	102.63	93.5
6	104.12	102.67	76.60	103.30	27.24	1.314	27.22	1.382	9.65	102.86	93.1
7	104.12	102.26	76.01	103.75	27.70	1.823	27.72	1.843	9.50	102.69	92.3
8	103.80	101.50	76.69	103.44	26.52	2.039	26.64	2.179	9.70	102.34	91.3
9	102.94	100.86	76.56	102.53	26.07	1.959	26.05	2.014	9.30	101.90	92.3
10	103.08	100.36	75.70	102.49	26.29	2.534	26.54	2.652	10.30	101.08	90.0
11	102.72	99.50	76.47	102.04	25.57	2.992	25.57	3.106	10.65	100.48	88.9
12	102.35	99.36	76.47	102.17	25.79	3.355	25.74	3.422	11.00	100.37	88.0
13	103.35	100.30	76.20	102.90	26.77	3.627	26.78	3.338	11.15	100.34	88.2
14	104.53	100.84	75.98	103.58	27.61	3.717	27.63	3.703	11.90	101.16	88.1
15	104.26	100.13	75.52	102.74	27.43	3.717	27.43	3.923	12.05	100.26	87.6
16	103.85	99.77	74.93	102.53	27.33	3.399	27.72	3.939	12.20	99.77	87.2
17	103.53	99.18	74.52	101.95	27.29	3.944	27.26	4.172	12.30	99.72	86.8
18	103.17	98.68	74.66	101.69	27.20	4.125	27.02	4.307	12.35	99.37	86.5
19	103.17	98.41	73.47	101.36	27.33	4.306	27.26	4.503	12.75	97.39	86.2
20	103.09	98.23	70.58	101.49	31.01	5.594	30.97	5.504	14.40	96.93	85.3
21	103.80	97.59	70.08	100.90	30.86	5.666	30.64	5.638	14.80	95.34	84.4
22	103.30	96.73	69.17	100.15	31.34	6.040	31.33	6.200	16.15	93.92	83.6
23	103.17	95.62	68.08	98.86	30.70	6.838	30.69	6.784	16.45	91.59	83.7
24	103.53	93.92	66.13	98.36	31.55	9.429	31.64	9.519	18.35	90.30	80.5
25	103.17	92.28	66.52	97.72	31.51	10.653	31.51	10.791	19.32	88.35	79.0
26	103.35	91.07	66.00	97.23	31.23	11.931	31.23	12.055	20.15	87.11	77.3
27	102.72	90.57	65.73	96.37	30.27	12.602	30.50	12.876	20.45	85.75	76.9
28	100.90	87.70	66.40	94.55	27.10	12.780	27.12	12.990	19.65	83.90	75.1
29	99.50	85.90	66.20	93.00	26.10	12.950	26.95	13.270	19.15	82.53	68.5

*Directly measured temperature difference.

Table [illegible]. Heat Transfer Data
(Cooling Water Flowing Upward)

No. of Run	$\dot{m}_c$ lbm/sec	Q_c Btu/hr	U Btu/hr-ft² F	h_1 Btu/hr-ft² F	St	μ_{1n} lbm/ft-sec	μ_{2n} lbm/ft-sec	G_c lbm/hr-sq ft	Pr	j	Re
1	0.02445	2700	123	133	0.00225	0.000442	0.000463	59300	4.66	0.00625	343
2	0.03275	3550	163	180	0.00227	0.000443	0.000470	79400	4.75	0.00635	453
3	0.04110	4260	186	211	0.00212	0.000444	0.000472	99600	4.82	0.00600	565
4	0.04940	5070	195	224	0.00188	0.000445	0.000434	119000	4.89	0.00535	662
5	0.05810	5600	196	225	0.00160	0.000445	0.000490	141000	4.96	0.00459	770
6	0.06630	6230	207	240	0.00149	0.000445	0.000495	161000	4.99	0.00429	870
7	0.07470	7060	214	250	0.00133	0.000445	0.000499	181000	5.02	0.00398	970
8	0.08130	7240	214	250	0.00127	0.000447	0.000503	197100	5.09	0.00370	1048
9	0.09230	8860	274	335	0.00149	0.000450	0.000499	224000	5.02	0.00431	1190
10	0.11050	9650	269	325	0.00121	0.000453	0.000502	268000	5.15	0.00356	1429
11	0.12900	10720	239	355	0.00113	0.000454	0.000517	313000	5.23	0.00334	1620
12	0.14650	12110	316	500	0.00112	0.000454	0.000522	355000	5.30	0.00334	1820
13	0.16650	14350	369	490	0.00121	0.000454	0.000520	404000	5.27	0.00360	2030
14	0.18610	16400	395	530	0.00119	0.000453	0.000521	447000	5.29	0.00355	2295
15	0.20250	17900	426	590	0.00120	0.000454	0.000525	491000	5.31	0.00353	2500
16	0.22100	19900	468	680	0.00127	0.000460	0.000526	536000	5.32	0.00330	2725
17	0.23900	21210	495	740	0.00123	0.000460	0.000530	580000	5.36	0.00334	2930
18	0.25900	22400	520	780	0.00124	0.000464	0.000531	623000	5.39	0.00334	3165
19	0.27650	24810	559	830	0.00131	0.000468	0.000532	670000	5.40	0.00396	3375
20	0.32650	32630	650	1130	0.00143	0.000471	0.000538	792000	5.46	0.00435	3940
21	0.37300	36900	717	1310	0.00147	0.000480	0.000545	895000	5.52	0.00451	4440
22	0.46600	46500	823	1750	0.00155	0.000490	0.000550	1130000	5.60	0.00472	5500
23	0.55700	53400	931	2220	0.00157	0.000502	0.000564	1350000	5.75	0.00496	6510
24	0.64000	61600	964	2420	0.00156	0.000510	0.000573	1552000	5.34	0.00497	7250
25	0.73700	68300	1021	2700	0.00151	0.000521	0.000583	1733000	5.97	0.00489	8200
26	0.82200	74700	1062	3000	0.00149	0.000526	0.000593	2020000	6.06	0.00486	9100
27	0.91200	78100	1095	3250	0.00147	0.000536	0.000600	2210000	6.15	0.00486	9850
28	1.01000	77800	1136	3550	0.00145	0.000550	0.000611	2450000	6.23	0.00486	10550
29	1.09000	77600	1162	3900	0.00147	0.000560	0.000670	2645000	6.95	0.00522	10550

Table [illegible]. Heat Transfer Date-Temperatures
(Cooling Water Flowing Downward)

o. of Run	t_{hi} F	t_{ho} F	t_{ci} F	t_{co} F	$\overline{t_{hi}-t_{ci}}^{*}$ F	$\overline{t_{ho}-t_{co}}^{*}$ F	Δt_a F	Δt_b F	Δt_{LMD} F	t_{1n} F	t_{2n} F
1	103.67	103.71	77.56	103.40	25.70	0.636	25.91	0.473	6.40	103.29	98.50
2	103.62	103.62	77.78	103.26	25.20	0.636	25.52	0.493	6.45	103.09	97.25
3	103.62	103.53	78.65	103.03	24.20	0.727	24.53	0.638	6.57	102.85	96.00
4	103.62	103.48	79.55	102.58	23.40	1.045	23.47	0.952	7.15	102.79	94.90
5	103.62	103.39	81.04	102.17	22.05	1.500	22.02	1.410	7.50	102.73	95.40
6	103.62	103.44	78.06	101.49	24.80	2.000	25.13	1.975	9.05	102.54	93.20
7	103.30	103.49	76.02	101.17	26.20	2.130	26.54	2.205	9.70	102.43	92.20
8	103.62	103.31	76.47	100.59	26.60	2.780	26.88	2.700	10.55	102.13	91.60
9	103.53	103.03	77.15	99.72	25.60	3.240	25.99	3.295	10.95	101.97	90.10
10	103.39	103.44	77.12	99.77	25.60	4.000	26.09	3.355	11.50	102.12	89.40
11	103.62	103.03	77.15	98.64	26.20	4.500	26.34	4.470	12.25	101.58	88.10
12	103.62	103.03	77.33	98.18	26.10	4.360	26.20	4.355	12.65	101.34	87.75
13	103.62	102.90	77.65	98.00	26.20	4.810	26.27	4.855	12.68	101.04	87.50
14	103.67	102.35	76.65	98.36	26.60	4.820	26.65	4.515	12.35	100.72	88.60
15	103.62	104.53	76.47	98.09	26.05	4.520	27.00	4.515	12.55	100.27	87.50
16	103.62	102.19	75.41	98.18	27.35	4.500	27.83	4.405	12.75	99.23	87.80
17	103.42	102.40	77.65	94.04	24.60	3.250	25.47	3.305	10.90	96.71	88.50
18	103.62	102.07	74.22	93.32	25.30	5.220	26.00	3.240	10.93	99.11	89.10
19	103.62	102.17	72.70	90.36	24.50	5.270	25.71	3.290	11.10	96.00	86.75
20	103.08	101.31	76.83	97.96	26.20	3.310	26.18	3.330	11.10	97.74	87.00
21	103.62	101.79	77.02	97.96	24.10	3.390	27.35	3.560	11.70	97.36	86.30
22	103.62	100.71	74.24	96.55	28.75	1.10	25.02	4.195	12.30	95.62	85.25
23	103.76	100.50	73.39	95.76	29.70	4.360	30.10	4.250	13.35	94.40	84.30
24	103.62	99.68	72.34	93.83	30.75	5.300	31.04	5.760	14.95	92.50	83.10
25	103.62	97.09	71.25	92.52	31.20	6.510	31.74	6.440	15.90	91.21	81.70
26	103.17	96.27	69.15	90.93	30.10	7.310	30.52	7.300	17.15	89.52	81.10
27	103.31	97.37	68.54	92.75	31.55	8.120	33.93	8.010	17.90	88.34	80.40

*Directly measured temperature difference.

Table [illegible]. Heat Transfer Data
(Cooling Water Flowing Downward)

No. of Run	$\dot{m}_c$ lbm/sec	Q_c Btu/hr	U Btu/hr-ft² F	h_1 Btu/hr-ft² F	St	μ_{1n} lbm/ft-sec	μ_{2n} lbm/ft-sec	G_c lbm/hr-sq ft	Pr	j	Re
1	0.02410	2230	100	106	0.00131	0.000443	0.000465	53500	4.69	0.00503	336
2	0.03230	2975	132	143	0.00130	0.000445	0.000470	79500	4.78	0.00504	452
3	0.04200	3650	160	179	0.00126	0.000445	0.000479	101900	4.32	0.00497	570
4	0.05070	4200	169	190	0.00114	0.000446	0.000434	123000	4.90	0.00435	690
5	0.05810	4890	169	189	0.00134	0.000446	0.000480	141000	4.84	0.00379	785
6	0.06650	5600	178	201	0.00125	0.000446	0.000492	161100	4.96	0.00359	900
7	0.07560	6530	195	225	0.00123	0.000447	0.000493	183200	5.02	0.00354	991
8	0.08400	7290	198	227	0.00118	0.000448	0.000501	203500	5.03	0.00325	1090
9	0.09220	7520	197	227	0.00101	0.000450	0.000511	223500	5.15	0.00296	1170
10	0.11450	9020	225	265	0.00096	0.000448	0.000515	277500	5.20	0.00281	1442
11	0.12920	10000	234	280	0.00090	0.000451	0.000522	313500	5.27	0.00266	1610
12	0.14800	11240	256	303	0.00086	0.000451	0.000524	359000	5.31	0.00256	1730
13	0.16650	12500	284	350	0.00087	0.000453	0.000525	403000	5.32	0.00259	2060
14	0.18530	14400	335	430	0.00094	0.000454	0.000520	457000	5.27	0.00279	2320
15	0.20350	15900	364	483	0.00093	0.000456	0.000525	493000	5.32	0.00293	2515
16	0.22300	13300	412	570	0.00105	0.000461	0.000524	541000	5.24	0.00311	2760
17	0.24050	13650	491	720	0.00124	0.000460	0.000520	583000	5.26	0.00369	3000
18	0.26300	20900	549	860	0.00132	0.000462	0.000516	650000	5.23	0.00391	3270
19	0.27700	22050	570	900	0.00134	0.000463	0.000523	671500	5.31	0.00401	3440
20	0.32400	25250	653	1140	0.00145	0.000470	0.000529	785000	5.36	0.00436	3980
21	0.37000	29010	722	1360	0.00152	0.000471	0.000530	897000	5.37	0.00457	4520
22	0.46500	37400	939	1800	0.00160	0.000480	0.000540	1127000	5.46	0.00487	5530
23	0.55500	44200	915	2140	0.00159	0.000486	0.000545	1345000	5.51	0.00489	6600
24	0.66500	51750	995	2530	0.00160	0.000498	0.000551	1612000	5.62	0.00498	7820
25	0.74800	57500	1038	2800	0.00155	0.000502	0.000564	1815000	5.74	0.00488	8600
26	0.82300	63000	1061	3000	0.00150	0.000515	0.000575	1995000	5.87	0.00472	9230
27	0.92600	69500	1114	3750	0.00167	0.000520	0.000590	2245000	5.91	0.00532	10250

Table 8 Results of Local Heat Transfer Coefficients Computation, (Run Number 14, cold water flowing up)

Location No.	t_{2x} °F (1)	t_{bcx} °F (2)	$(\Delta t_{bc})x$ °F (3)	q_x Btu/hr.11" (4)	$(t_{bh}-t_1)x$ °F (5)	$(t_1-t_{bc})x$ °F (6)	h_x Btu/hr.ft.^{2}°F (7)
0		98.36					
			0.26	174	0.38	4.1	157
1	95.5	98.1					
			0.3	201	0.44	4.5	160
2	94.5	97.8					
			0.5	334	0.73	4.9	253
3	94.5	97.3					
			0.4	268	0.59	5.5	181
4	93.2	96.6					
			1.0	669	1.46	5.8	428
5	91.5	95.6					
			1.4	936	2.05	6.5	535
6	90.1	94.2					
			2.0	1340	2.94	7.2	690
7	88.3	92.2					
			2.6	1740	3.81	8.8	733
8	86.5	89.6					
			2.6	1740	3.81	11.0	586
9	84.2	87.0					
			2.5	1670	3.66	13.7	451
10	81.9	84.5					
			2.7	1800	3.94	16.5	404
11	78.6	81.8					
			2.7	1800	3.94	19.5	342
12	77.4	79.1					
			2.22	1485	3.26	22.1	249
13		76.88					

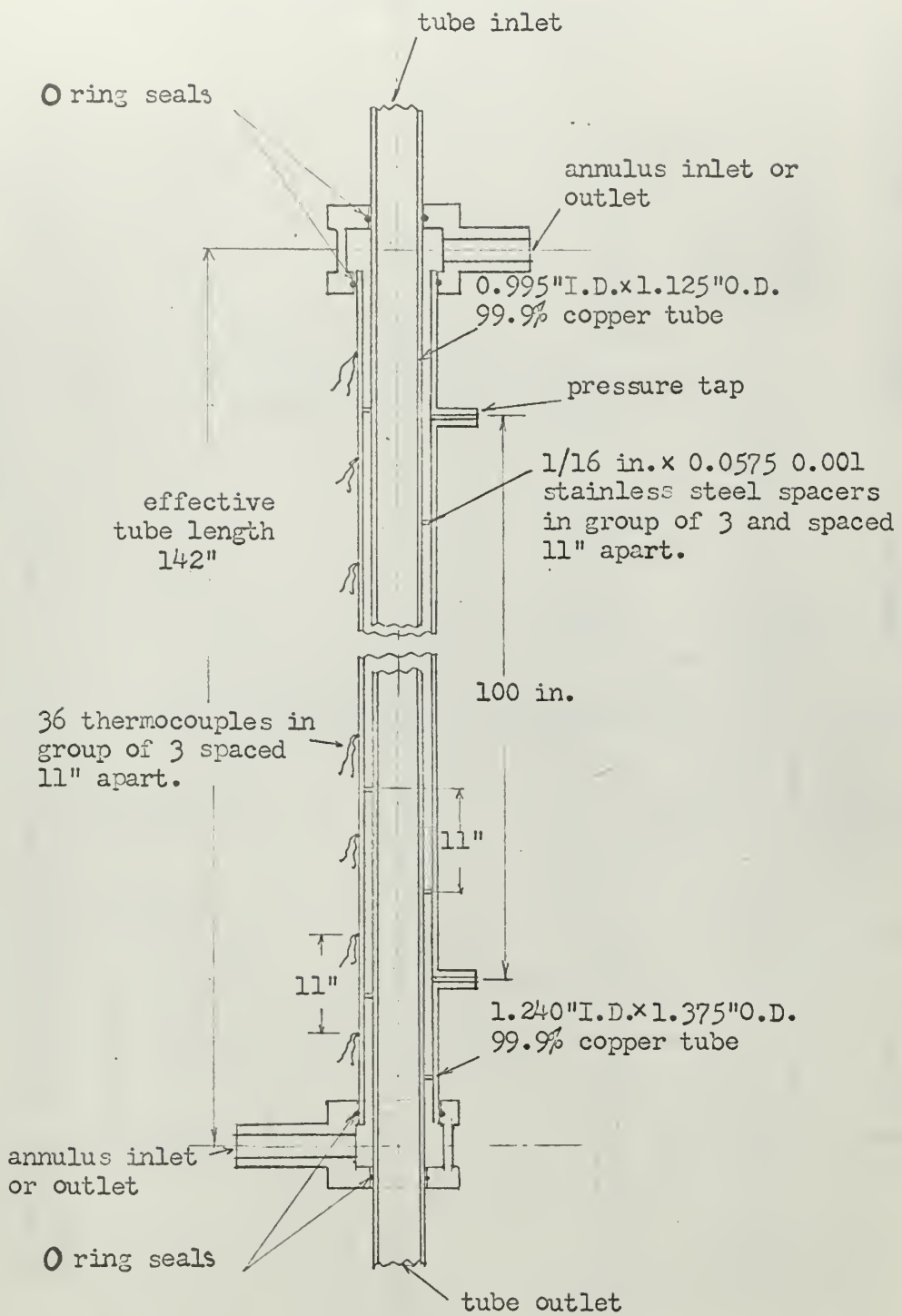

tube inlet
O ring seals
annulus inlet or outlet
0.995"I.D.x1.125"O.D.
99.9% copper tube
pressure tap
1/16 in.x 0.0575 0.001 stainless steel spacers in group of 3 and spaced 11" apart.
effective tube length 142"
100 in.
36 thermocouples in group of 3 spaced 11" apart.
11"
11"
1.240"I.D.x1.375"O.D.
99.9% copper tube
annulus inlet or outlet
O ring seals
tube outlet

Figure 1

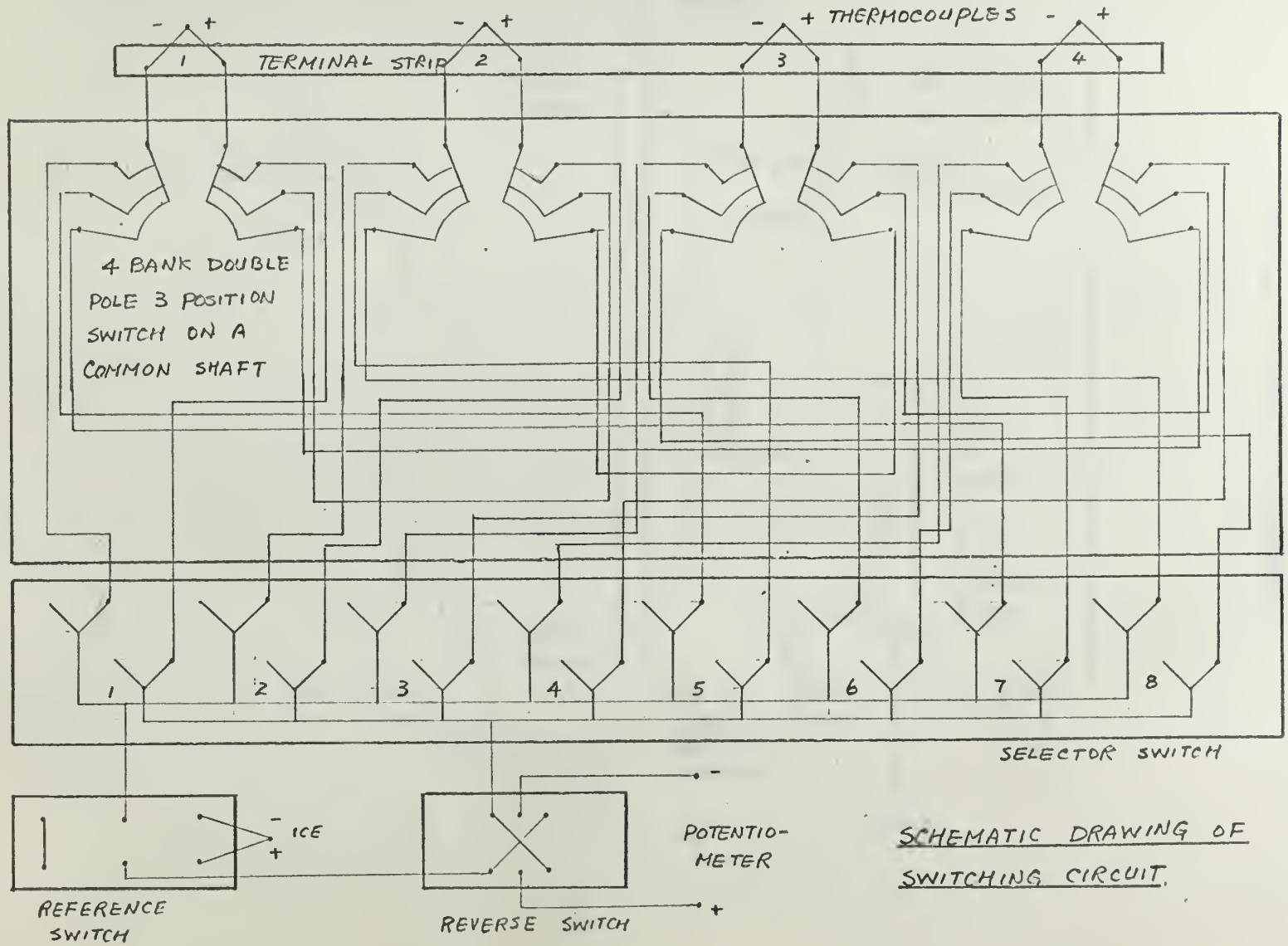
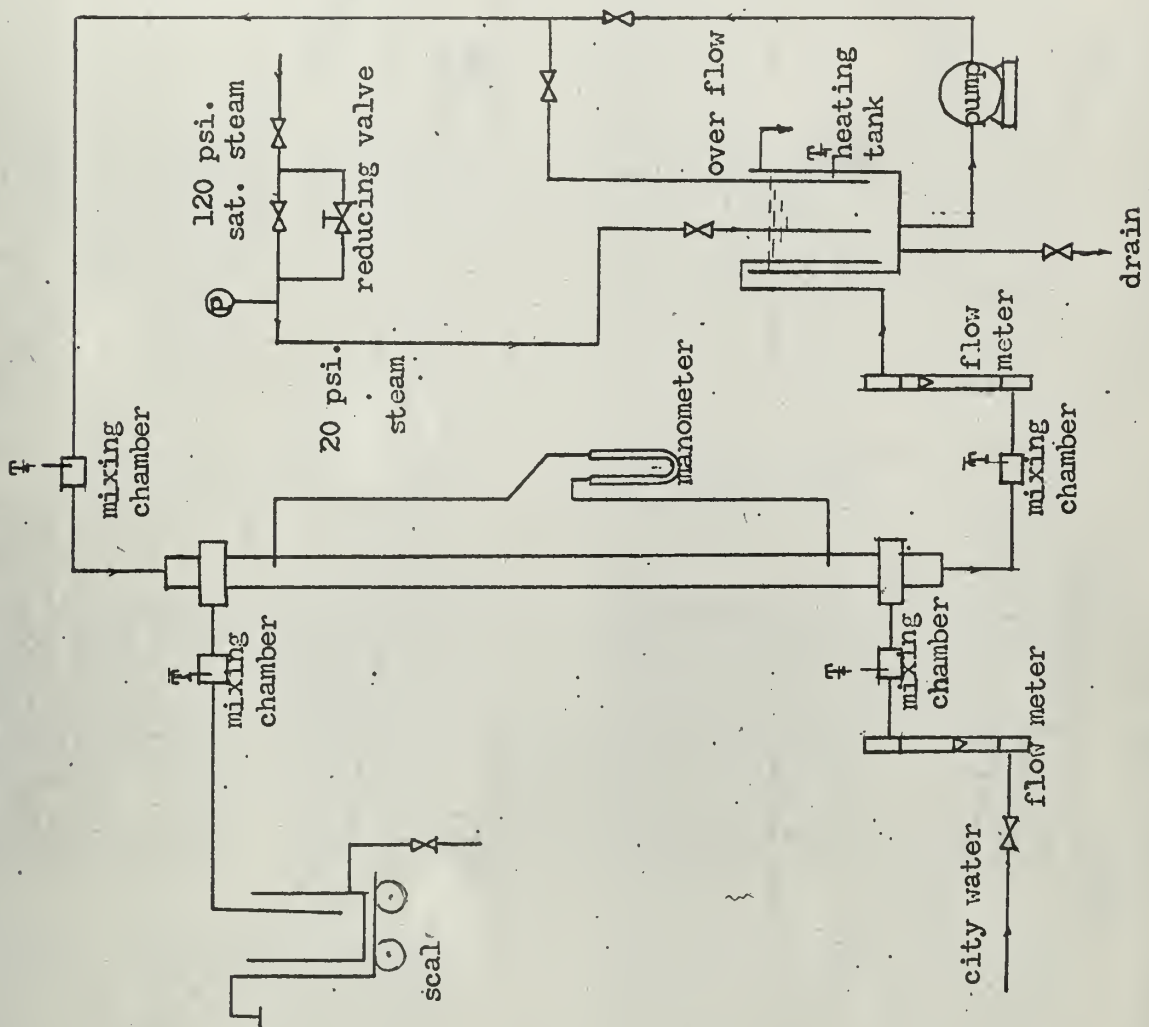


Figure 2

SCHEMATIC DRAWING OF SWITCHING CIRCUIT.

SCHEMATIC ARRANGMENT OF EQUIPMENT



Figures 3

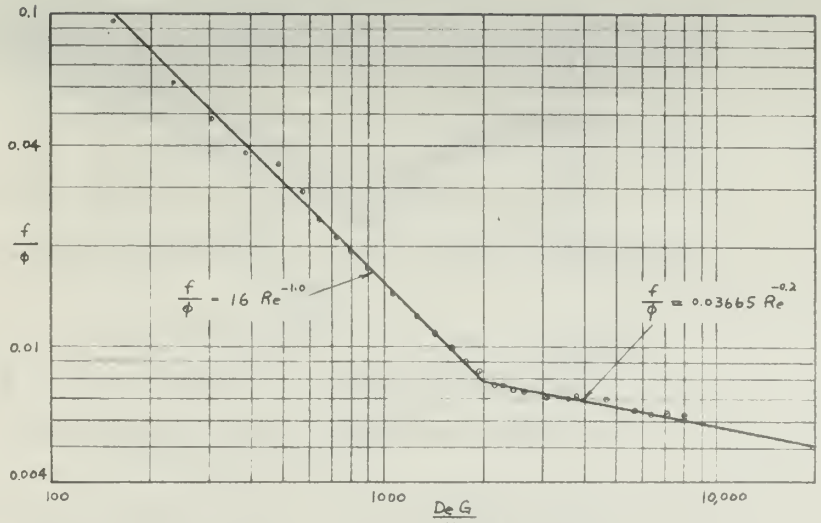


Figure 4 Isothermal Friction Data, Cooling Water Flowing Up.

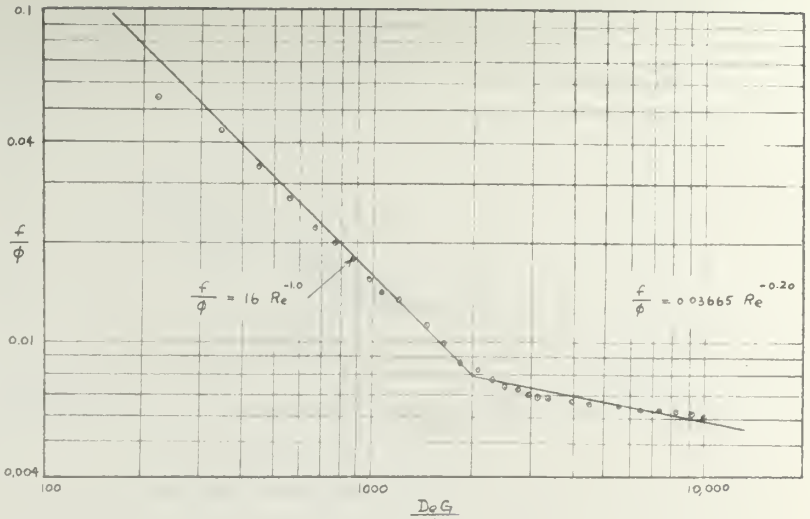


Figure 6 Internally Heated Friction Data, Cooling Water Flowing Up.

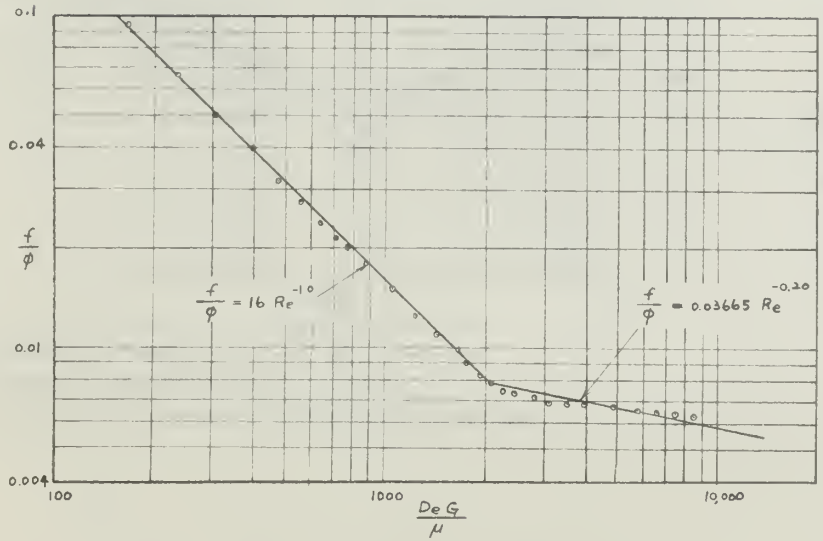


Figure 5 Isothermal Friction Data, Cooling Water Flowing Down.

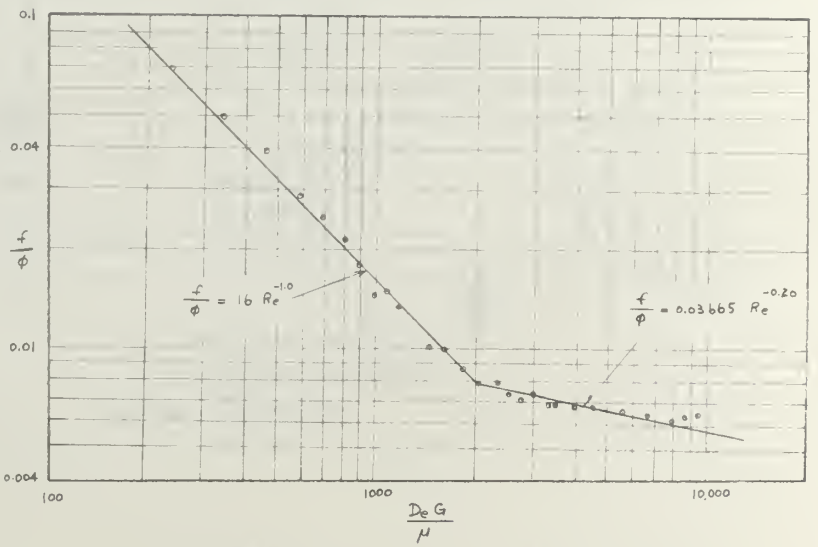
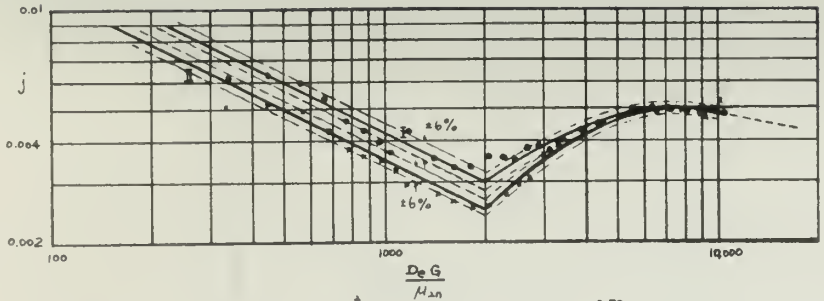


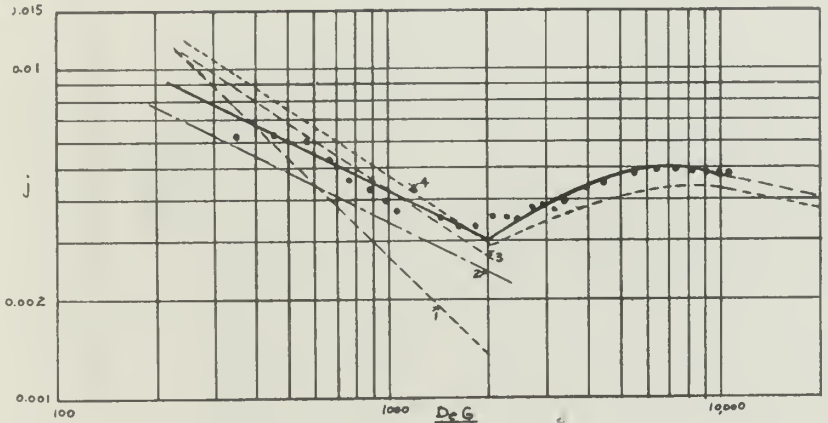
Figure 7 Internally Heated Friction Data, Cooling Water Flowing Down.



I. $j = \left(\frac{h}{G C_p}\right)\left(\frac{C_p \mu}{k}\right)^{\frac{2}{3}}_{an}\left(\frac{\mu_{in}}{\mu_{an}}\right)^{0.14} = 0.132\,(Re)^{-0.50}_{an}$, For Cooling Water Flowing Up.

II. $j = 0.1065\,(Re)^{-0.50}_{an}$, For Cooling Water Flowing Down.

Figure 8 Heat Transfer Coefficients for Annular Space, Correlated in Terms of Average Outer Wall Temperatures.

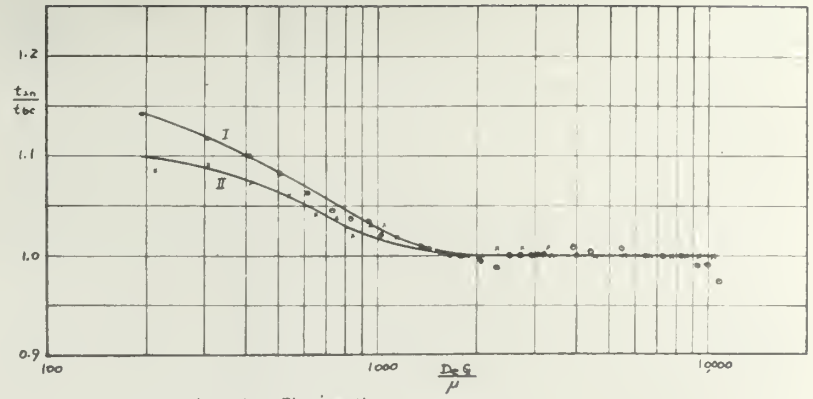


Curve 1. Jakob and Rees, [4]. Curve 2. Burt, [8]

3. Norris and Streid, (Inside Tube, $\frac{L}{D_2 + D_1} = 60$), [11]

4. Sieder and Tate, (Inside Tube, $\frac{L}{D_2 + D_1} = 60$), [6]

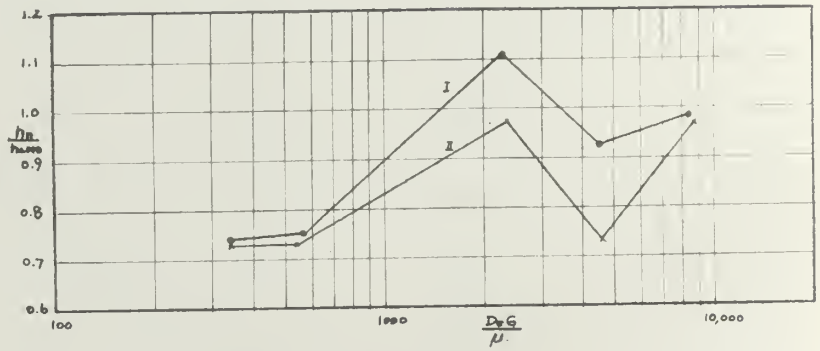
Figure 12 Comparison of Heat Transfer Data, Upward Flowing Only.



I Cold Water Flowing Up.

II Cold Water Flowing Down.

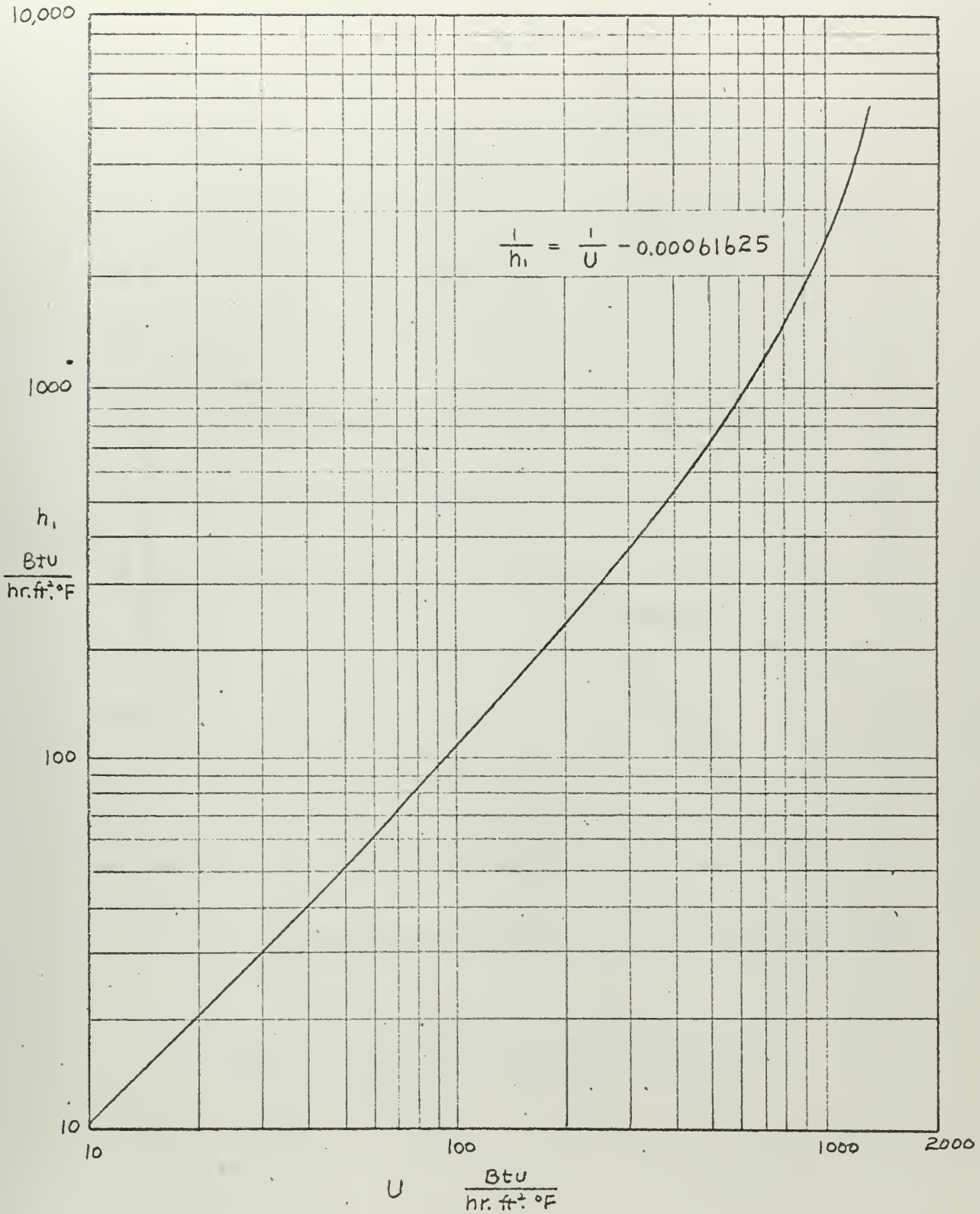
Figure 11 t_{2n}/t_{bc} versus Re



I. Cooling Water Flowing Up.

II Cooling Water Flowing Down.

Figure 13 h_n/h_{LMD} versus Re



Figrue 9 h_1 Value Evaluating Chart

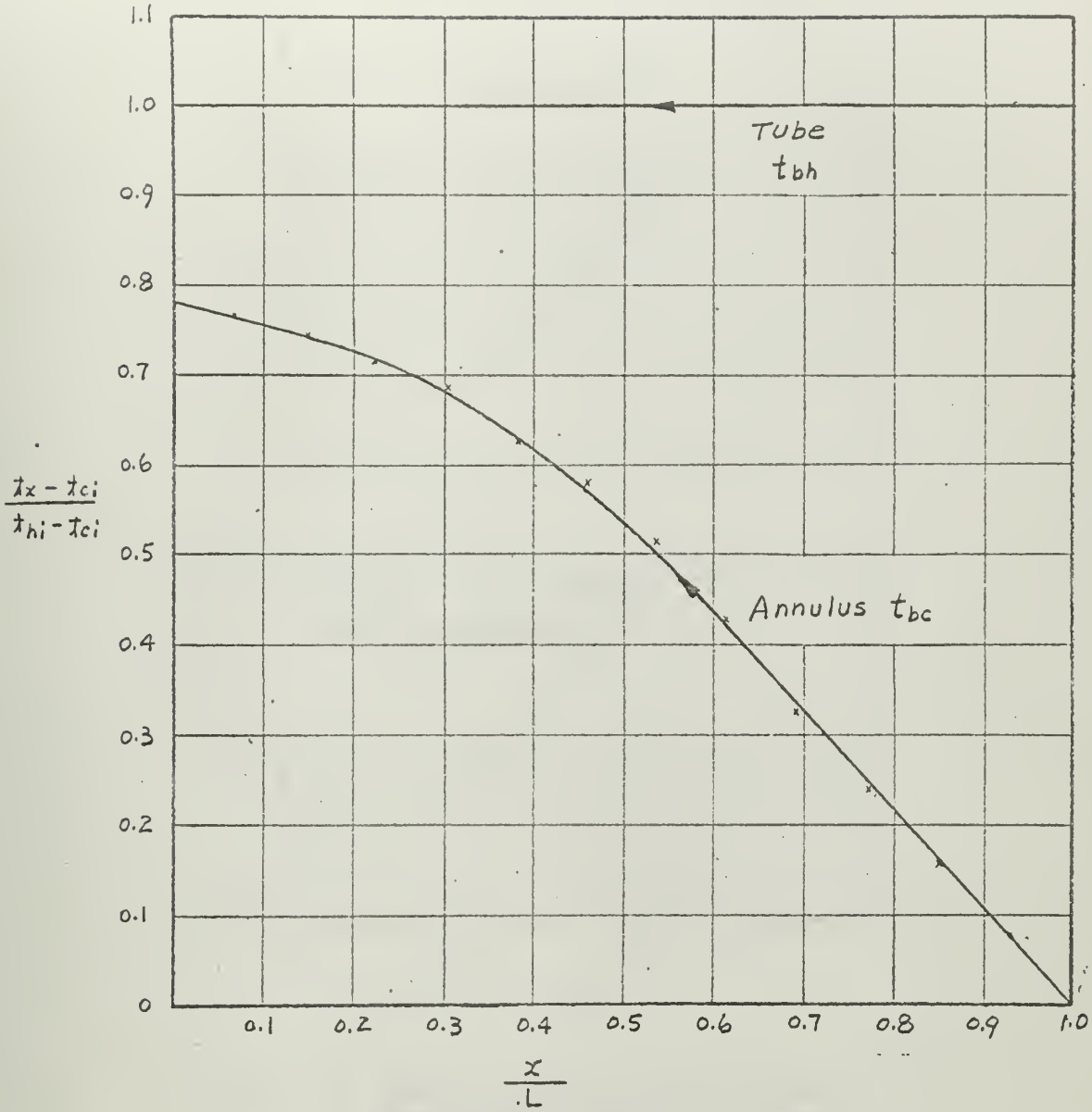


Figure 10 Outer Wall Temperature Variation of Annulus

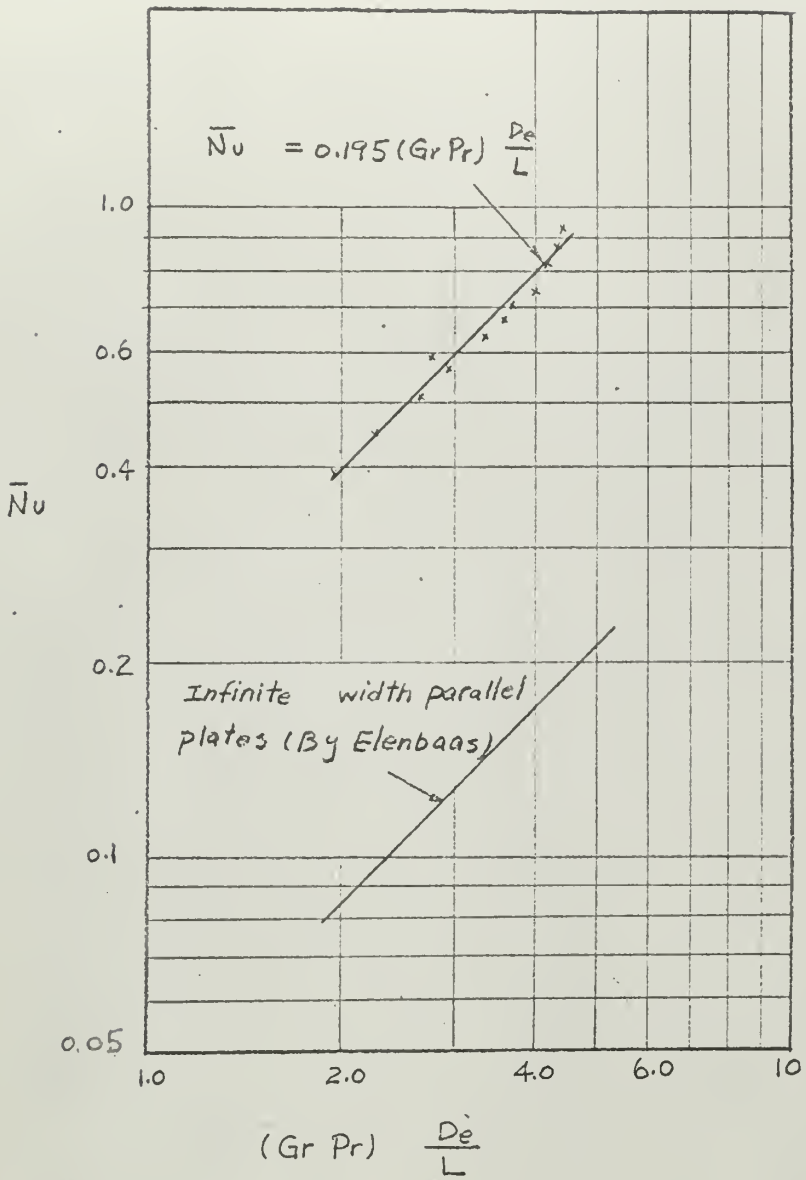


Figure 14 Natural Convection Heat Transfer Coefficients for Annular Space, Correlated in Terms of Average Outer Wall Temperatures.

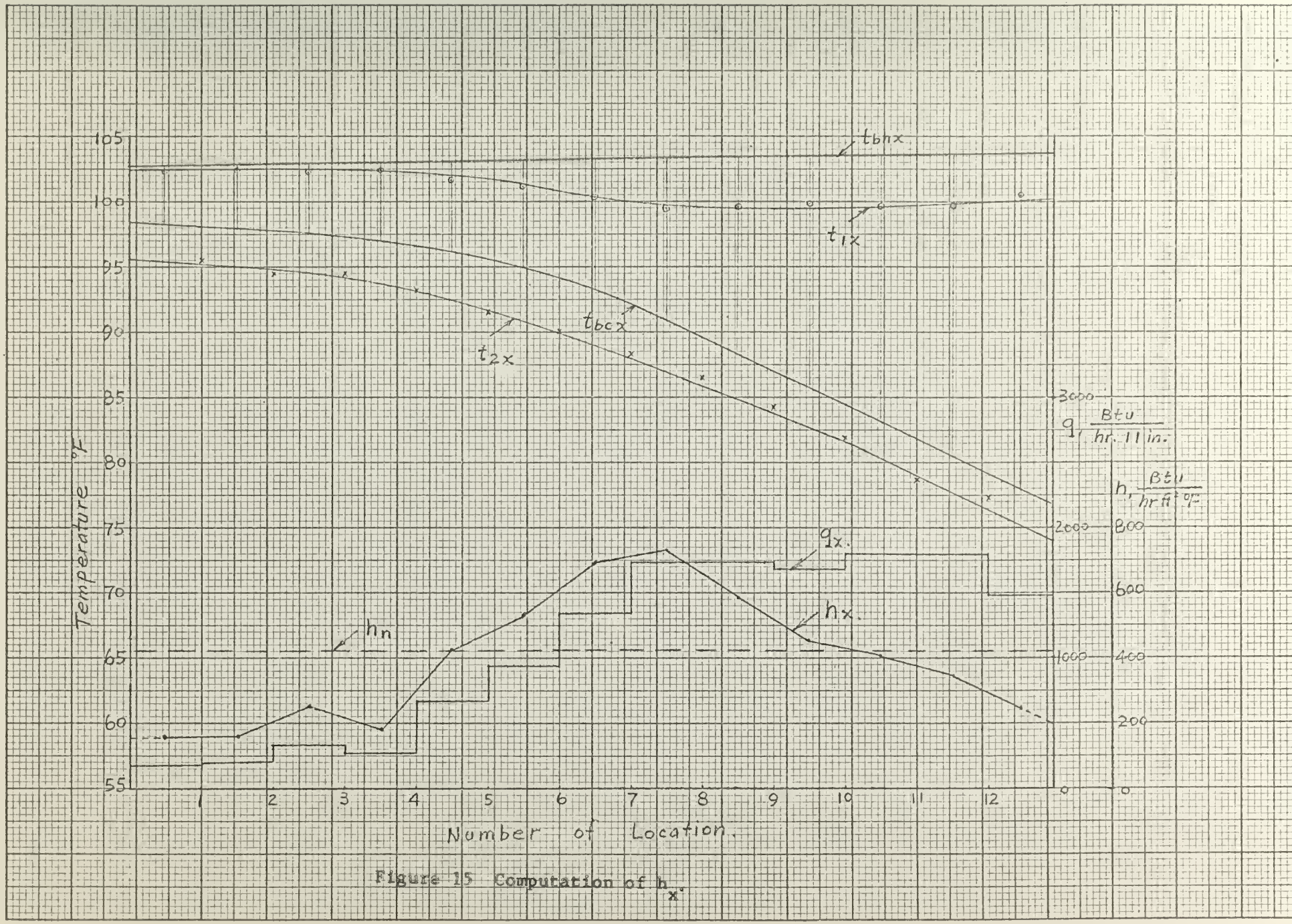


Figure 15 Computation of h_x.

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Pressure drop and heat transfer in a ver
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